Renewable and Sustainable Energy Reviews 41 (2015) 965–974

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Pressure fluctuations in the vaneless space of High-head pump-turbines—A review

Zhigang Zuo*, Shuhong Liu, Yuekun Sun, Yulin Wu

State Key Laboratory of Hydro science and Engineering, Department of Thermal Engineering, Tsinghua University, Beijing, China

ARTICLE INFO

Article history:
Received 2 April 2014
Received in revised form
5 August 2014
Accepted 5 September 2014

Keywords:
Pressure fluctuations
Vaneless space
High-head pump-turbines
Rotor–stator interactions

ABSTRACT

In high-head pump-turbines, observations in the engineering field show that the most detrimental hydraulic instabilities are pressure fluctuations in the vaneless space. For example, these pressure fluctuations can cause vibrations and fatigue failures in hydraulic components. It has been recognized that pressure fluctuations in vaneless space are induced primarily by the interactions between runner blades and guide vanes, i.e., rotor–stator interactions (RSI).

The present paper presents and discusses studies in this field, which are carried out by various investigators. Studies include mechanism research on RSI, experimental and numerical investigations on RSI characteristics, discussions on the relationship between vibrations and pressure fluctuations, studies on the geometric and operating parameters of the unit that influence the pressure fluctuations in vaneless space, and precautions and countermeasures to reduce these pressure fluctuations.

Contents

1. Introduction

Hydropower pumped storage has been recognized as the only commercially proven grid-scale energy storage technology. During off-peak periods, water is pumped to an upper storage reservoir to generate hydroelectric power to cover temporary peaks in demand. In the 1960s there was a period of rapid development of pumped storage as a complement to nuclear power [1]. In the recent decades, this technology has been used extensively, especially with the increasingly renewable-based electricity supply systems, for the benefits of load-leveling, grid frequency regulation, and spinning reserves [2]. By 2013 there was a reported total

* Corresponding author.
E-mail address: zhigang200@tsinghua.edu.cn (Z. Zuo).

http://dx.doi.org/10.1016/j.rser.2014.09.011
1364-0321/

Nomenclature

- D diameter of runner
- D_0 pitch diameter of guide vanes opening
- D_4 inner diameter of guide vanes
- D_d diameter of runner
- E specific hydraulic energy of the unit
- f_1 torque transducer on the shaft
- f_n rotating frequency of runner
- g gravitational acceleration
- H hydraulic head
- k number of diametrical nodes of vibration
- m arbitrary integer
- n arbitrary integer, rotating speed of the unit
- n_{ED} speed factor, $n_{ED} = \frac{nD}{\sqrt{E}}$
- n_q specific speed at pump mode, $n_q = \frac{n\sqrt{Q}}{H^{3/4}}$
- $p_1 \sim p_7$ pressure transducers on the downstream side of the draft tube cone, on the upstream side of the draft tube cone, at the spiral case inlet, in the draft tube cone, in the vaneless space, along the intake, in the draft tube outlet, respectively
- Q volumetric flow rate
- Q_{ED} discharge factor, $Q_{ED} = \frac{Q}{D^2\sqrt{E}}$
- t_1 axial and redial thrust transducers on the shaft-runner coupling flange
- T torque
- T_{ED} torque factor, $T_{ED} = \frac{T}{\rho D^3 E}$
- Z_g number of guide vanes
- Z_r number of runner blades
- $\frac{\Delta H}{H}$ relative amplitude of pressure fluctuation
- ΔH total amplitude of pressure fluctuations in terms of hydraulic head
- θ_r angle of runner rotation
- θ_h angle of travelling of hydraulic impacts in stationary coordinates
- ψ head coefficient, $\psi = \frac{gH}{D^2 n^2}$
- ϕ flow coefficient, $\phi = \frac{Q}{D^3 n}$
- ϕ_h angle of travelling of hydraulic impacts in rotating coordinates
- ω angular velocity of runner

capacity of more than 127 GW of pumped storage power stations (PSPS) worldwide [3].

The first pumped storage power station in the world appeared in the Alpine regions of Switzerland in the 1890s. Early designs used separate pump impellers and turbine runners. With the development of hydraulic research, reversible pump-turbines, or pumped-storage turbines, became the dominant design in the 1960s, due to their relatively compact sizes [4]. In recent years, higher heads and larger unit capacities have been adopted to further reduce manufacturing and construction costs [5–16]. Fig. 1 demonstrates the upward trend of pumping head of PSPS over the years.

In high-head pump-turbines, the problems of hydraulic instabilities, i.e., pressure fluctuations within the unit, have become more prominent. Pressure fluctuations may induce mechanical vibrations, and in some cases premature mechanical failures. Egusquiza et al. [17] reported a failure investigation regarding a large pump-turbine runner (400 m head, maximum power of 110 MW), induced by pressure fluctuations. A part of the runner crown broke off during operation, and caused further damage when passing through the machine (see Fig. 2). Fig. 3 shows a close-up of the crack. Clear beach marks reveal a fatigue failure. Further studies indicated that pressure fluctuations in vaneless space contributed greatly to this failure.

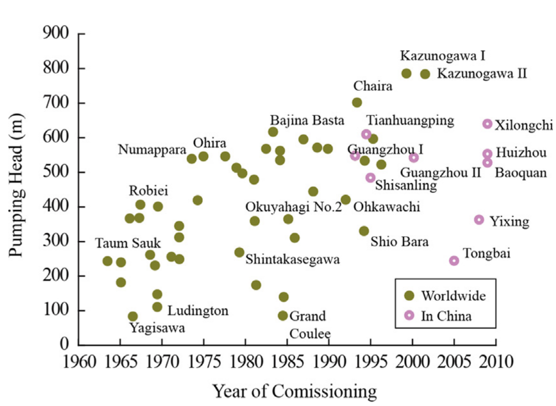

Fig. 1. Development of head of PSPS, data collected from Refs. [5–16].

In fact, in high-head pump-turbines, it has been observed that the pressure fluctuations in the vaneless space are the most detrimental hydraulic instabilities, as shown in Table 1 [6–16].

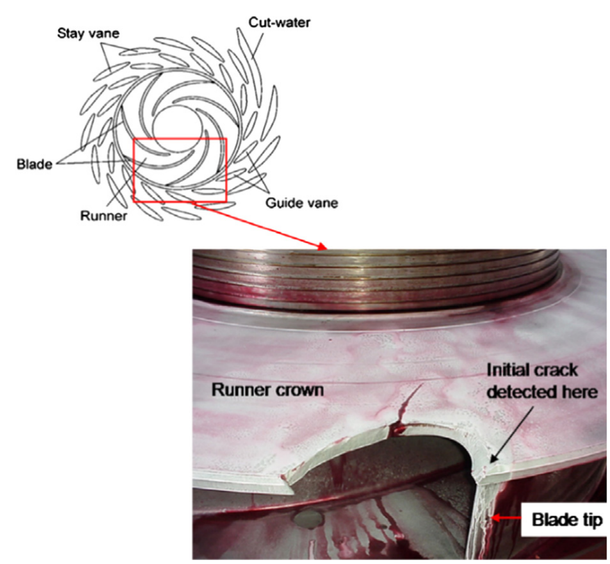

Fig. 2. Sketch of the runner and distributor and a picture of the broken runner [17].

Fig. 3. Crack on the runner with beach marks [17].

Table 1
Pressure fluctuations in different PSPS in China.

PSPS	Guangzhou I	Shisanling	Tianhuangping	Yixing	Xilongchi	Huizhou	Baoquan
No. of stay vanes, guide vanes, and runner blades	20/20/7	16/16/7	26/26/9	26/26/9	20/20/7	20/20/9	20/20/9
Year of commissioning	1993	1995	1998	2008	2009	2009	2009
Manufacturer of pump-turbines	Cegelec(al)	Voith & ELIN	Kvaerner	GE (Norway)	Toshiba Hitachi Mitsubishi	Alstom dec	Alstom
Operating head range (m)	496–536 (T) 514–550 (P)	428–475 (T) 442–490 (P)	512–607 (T) 532–615 (P)	344–407 (T) 360–420 (P)	612–688 (T) 634–703 (P)	496–554 (T) 515–565 (P)	500–565 (T) 498–575 (P)
Discharge (m^3/s)	68.37 (T) 60.03 (P)	53.8 (T) 37.8 (P)	67.7 (T) 57.7 (P, Max.)	79 (T) 71.5 (P, Max.)	55.88 (T) 36.84 (P, Min.)	67.4 (T) 56.54 (P, Max.)	68.8 (T) 55.2 (P, Max.)
Installed capacity (MW)	306 (T) 326.08 (P)	204 (T) 218 (P, Max.)	306 (T) 336 (P, Max.)	255 (T) 267.5 (P, Max.)	306 (T) 319.6 (P. Max.)	306 (T) 310 (P, Max.)	306 (T) 315.4 (P, Max.)
Specific speed (m, m^3/s)	35.7	29.6	31.2	40.7	27.1	34.4	35.2
Rotational speed (r/min)	500	500	500	375	500	500	500
Maximum pressure fluctuations	14.4%	16.3%	54.4% (runaway condition)	4.2%	11.3%	5.0%	9.52%
Location of max pressure fluctuations	Vaneless space	Vaneless space	Vaneless space	Vaneless space	Vaneless space	Vaneless space	Vaneless space

Pressure fluctuation data were collected from Refs. [6–16] and results of experimental reports.
T: Turbine mode.
P: Pump mode.

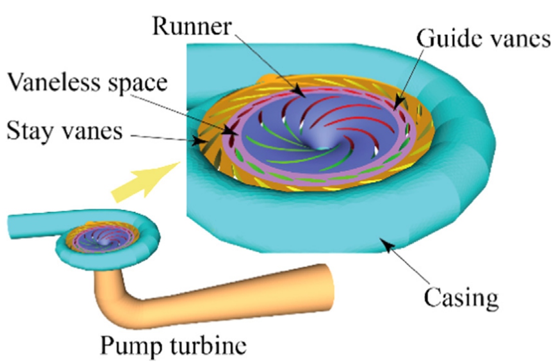

Fig. 4. Vaneless space between the runner and guide vanes in pump-turbines.

Vaneless space in pump-turbines refers to the gap between runner blades and guide vanes, as shown in Fig. 4. It is one of the major locations for pressure fluctuation monitoring. Fig. 5 shows the locations of pressure transducers for a pump-turbine in acceptance tests suggested by the International Electrotechnical Commission (IEC) standard, where $p_1 \sim p_7$ are pressure transducers on the downstream side of the draft tube cone, on the upstream side of the draft tube cone, at the spiral case inlet, in the draft tube cone, in the vaneless space, along the intake, and in the draft tube outlet, respectively. t_1 indicates the axial and redial thrust transducers on the shaft-runner coupling flange. f_1 indicates the torque transducer on the shaft [18].

In order to determine the flow mechanism of this phenomenon and to identify gaps for future studies the present paper discusses the engineering observations and studies. The researchers whose work is covered in this paper focused on pressure fluctuations in the vaneless space of high-head pump-turbines.

Previous related studies cover the following topics:

a. Mechanism studies on the rotor–stator interactions, by means of theoretical and experimental analysis, to clarify the source of pressure fluctuations in vaneless space.
b. Characteristics of pressure fluctuations in vaneless space, including the amplitude and main frequencies, using experimental and numerical methods.
c. Relationship between vibrations and pressure fluctuations in engineering applications and experimental tests, to investigate the consequence of pressure fluctuations on the prototype and model running.
d. Geometric and operating parameters of the unit that influence pressure fluctuations in vaneless space.
e. Precautions and countermeasures to reduce pressure fluctuations in vaneless space.

2. Mechanism studies on rotor–stator interactions

It has been recognized that pressure fluctuations in vaneless space are induced primarily by the interactions between runner blades and guide vanes, i.e., rotor–stator interactions (RSI). This RSI phenomenon can be considered as the combination of two mechanisms, potential flow (inviscid) interaction and wake (viscous) interaction. Dring et al. [19] concluded that for a gas turbine, the potential flow interaction can cause unsteadiness in both the upstream and downstream rows if the axial gap between them is less than approximately the airfoil chord. The reason for this relates to the fact that the gradients due to potential flow decay with a length scale of the pitch (or chord) of the cascade. Because the wake is convected downstream and decays much more gradually in the far field, wake interaction will be present even when adjacent rows are spaced far apart.

Through both theoretical and experimental studies, Tanaka [20] developed a model to determine the diametrical vibration modes in high-head pump-turbines. Since the thickness of the guide vanes is large for high-head pump-turbines, significant hydraulic excitation force is developed when the runner blades pass across the wakes of the vanes. This interference gives rise to vibrations in the runner with a principle frequency of $Z_g \times f_n$ and its high order harmonics of $n \times Z_g \times f_n$ while observed from rotating coordinates, and a principle frequency of $Z_r \times f_n$ (*Blade Passing Frequency, BPF*) and its high order harmonics of $m \times Z_r \times f_n$ while observed from stationary coordinates. Z_g is the number of guide vanes, Z_r is the number of runner blades, f_n is the rotational frequency of the runner, m and n are arbitrary integers. The vibration mode with k

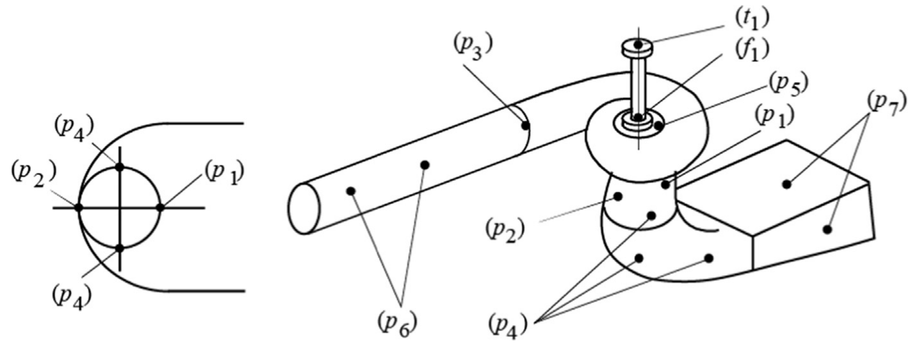

Fig. 5. Suggested locations of pressure transducers for a pump-turbine [18].

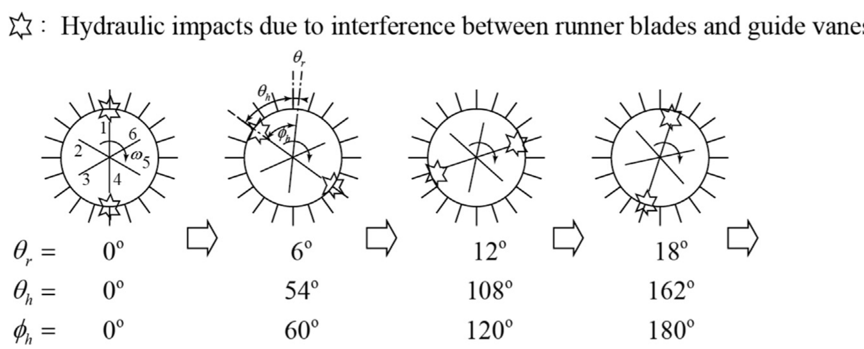

Fig. 6. Hydraulic interference of runner blades and guide vanes [20].

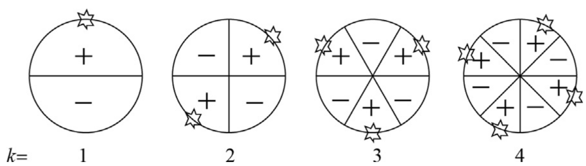

Fig. 7. Vibration modes with k diametrical nodes [20].

diametrical nodes can be determined using

$$n \times Z_g \pm k = m \times Z_r \tag{1}$$

Fig. 6 shows an example of interference in the case of $Z_g = 20$ and $Z_r = 6$. Runner blades 1 and 4, and runner blades 2 and 5 are sequentially excited by interference with the guide vane wakes. The in-phase excitations of runner blades 1 and 4 or 2 and 5 induce a vibration having a mode with 2 diametrical nodes. This vibration mode rotates in the direction opposite from the runner rotation. Fig. 7 shows the vibration modes with $k = 1 \sim 4$ diametrical nodes. Tanaka also pointed out that the mode of the pressure fluctuations could also be analyzed following the same logic. From the experiments it was seen that the natural frequency of the runner installed in water became less than 50% of that in air due to the added mass effect of the water.

Ruchonnet et al. [21] presented a numerical simulation of the RSI phenomenon based on a one-dimensional hydroacoustic model. Hydraulic parts were modeled as hydroacoustic components, e.g., resistances, inductances and capacitances, based on their acoustic characteristics for frequency analyses. A model high-head pump-turbine with 20 guide vanes and 9 runner blades was investigated. An analysis of the pressure fluctuations resulting from the rotor–stator excitation showed that RSI patterns, such as the rotating diametrical pressure mode and the standing wave, could be properly simulated. A similar approach has been adopted to discuss the influence of the thickness of guide vanes and runner blades, runner blade wave speed, guide vane wave speed and

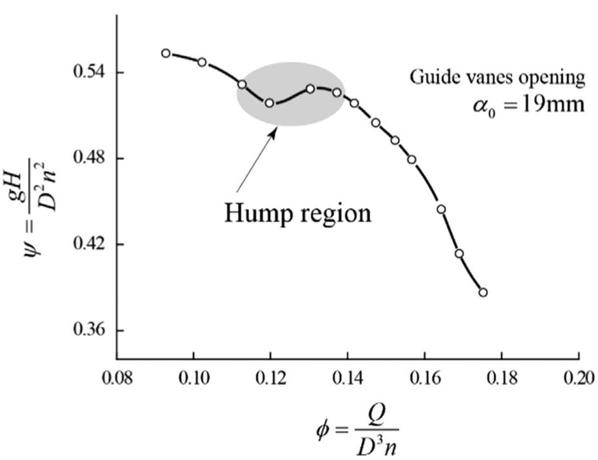

Fig. 8. Performance curve of the pump-turbine in pump mode. Data are from reference [29].

runner rotating frequency [22,23]. These parameters change the characteristics of the RSI, which in turn vary the pressure fluctuations.

Rodriguez et al. [24] presented a theoretical analysis based on the sequence of interaction to describe the characteristics in the frequency domain of vibrations originating from the RSI. This analysis also provided suggestions regarding the descriptions of amplitude and frequency of pressure fluctuations in vaneless space.

3. Experimental and numerical studies

Experimental tests on pressure fluctuations in pump-turbines have been carried out since 1970. Grein et al. [25] and Yamaguchi et al. [26] both measured the stress and pressure fluctuations in a prototype and a model pump-turbine. Similarities between the

data obtained in the model and in prototype tests were discussed in these two papers, both of which focused on the factors influencing the amplitude and frequency of pressure fluctuations at different positions along the flow paths.

Liu et al. [27] carried out more detailed experiments on different features of pressure fluctuations under different operating conditions. The results indicated that in pump mode, the amplitude of pressure fluctuations was at the minimum at the design operating point. The main frequency of pressure fluctuations in vaneless space was BPF, while the amplitude of pressure fluctuations was 30% larger under critical cavitation condition than under non-cavitating conditions. The article also discussed the pressure fluctuations at different openings of the guide vanes.

Regarding the stress and vibrations caused by the pressure fluctuations in vaneless space, Kawamoto et al. [28] were the first to discuss the method of reducing the amplitude of pressure fluctuations. Results in this article showed that by increasing the guide vane height by 40%, the amplitude of the pressure fluctuation at the runner periphery under turbine mode was reduced by 20–30%. This was an early attempt to reduce the stress fluctuations by reducing pressure fluctuations.

Wang et al. [29] carried out experimental studies to investigate the pressure fluctuations in the so-called ‘hump region’ on a pump-turbine's performance curve in pump mode. As shown in Fig. 8, in the hump region, a single value of *head coefficient* $\psi = (gH/D^2n^2)$ (H,D,g and n are the head, runner diameter, gravitational acceleration, and rotating speed of the unit) corresponds to multiple values of *flow coefficient* $\phi = (Q/D^3n)$ (Q is the volumetric flow rate), which can cause instability in the operations. The experiments were performed on HII, the second universal test rig for high-head hydraulic machinery in HILEM (*Harbin Institute of Large Electrical Machinery*). Results showed that the pressure fluctuations in vaneless space increased with increasing openings of the guide vanes. The hydraulic performance was increasingly unstable with larger guide vane openings in the hump region, consequently getting through the hump region by reducing the opening of guide vanes is often used for the unit start-ups in pump mode.

Hasmatuchi et al. [30] investigated the flow hydrodynamics in a reduced pump-turbine model under off-design operating conditions in generating mode. The investigation included wall pressure measurements in the stator, synchronized with high-speed flow visualizations in the vaneless space between the runner and the guide vanes using air bubble injection. The results showed that at the best efficiency point (BEP), the pressure fluctuations were very low and were dominated by BPF and its first harmonic. As the runner entered the S-shaped characteristic on the performance curves at the turbine mode, a substantial increase in the pressure fluctuation was observed. Fig. 9 illustrates the S-shaped characteristics in $Q_{ED} \sim n_{ED}$ curves, in which $Q_{ED} = (Q/D^2\sqrt{E})$ denotes the discharge factor, $n_{ED} = (nD/\sqrt{E})$ denotes the speed factor, $T_{ED} = (T/\rho D^3 E)$ denotes the torque factor

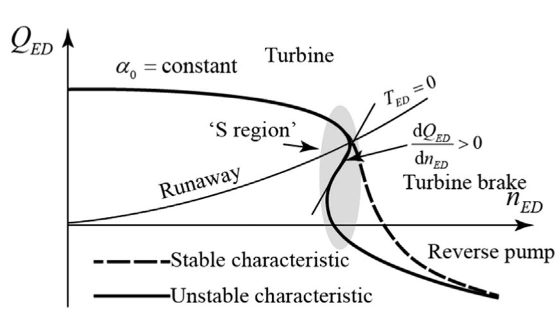

Fig. 9. S-shaped characteristic curves of a pump-turbine in turbine mode [18]. (a) Synchronous guide vanes (b) Misaligned guide vanes (MGV).

(E is the specific hydraulic energy of the unit, and T is the torque). The $(dQ_{ED}/dn_{ED}) > 0$ condition in this region may cause operational instabilities in transient processes, such as at unit start-up or runaway. At runaway, the spectral analysis showed an increase in the amplitude of a low frequency component at about 70% of the runner rotational frequency. The amplitude further increased as the zero discharge condition was approached. The time series of pressure distribution in the vaneless space between the runner and guide vanes revealed one stall cell rotating with the runner at a sub synchronous speed.

Ran et al. [31] analyzed the operating instabilities under large partial flow conditions for a pump-turbine. The model pump-turbine's hydraulic performance was tested with the pressure fluctuations measured at unstable operating points near a positive slope in the performance curve. The hydraulic performance tests showed two separate positive slope regions for the pump-turbine. The flow discharge for the first positive slope was 0.85 to 0.91 times that at the BEP. The pressure fluctuation amplitudes under these unstable large partial flow conditions near the first positive slope were much larger than during stable operations. A dominant frequency was measured at 0.2 times the runner rotational frequency in the flow passage near the runner exit, which was believed to be induced by the rotating stall in the flow passage of the wicket gates. The test results also showed hysteresis with pressure fluctuations when the pump-turbine was operated near the first positive slope, which created different pressure fluctuations for those operation points, even though their flow rates and heads were similar.

Unlike in model pump-turbines, pressure fluctuations due to RSI in installed pump-turbines are difficult to monitor, because the units are not manufactured with built-in pressure sensors. Rodriguez et al. [32] recently reported a study of RSI characteristics by the means of vibration measurements on a 100 MW pump-turbine. Two vibration sensors were installed rotating with the shaft instead of in bearings. This was done to avoid the effect of bearing response on the measurements. This technique is especially important because of the relatively low rotating speed and stiff shaft of large scale pump-turbines, which result in a much higher first rotor critical speed than the first bearing natural frequency. Thus lower harmonics of BPF in the vibrations measured in the bearing are more susceptible than ones measured rotating with the shaft. A comparison with pressure fluctuation measurements confirmed the validity of this method.

With the development of computer technology, *Computational Fluid Dynamics* (CFD) has become the primary method of investigating characteristics of pressure fluctuations in hydraulic machinery. CFD methods for unsteady flow simulations have been used to acquire amplitudes and frequencies of pressure fluctuations in pump-turbines, among which Reynolds Averaged Navier–Stokes (RANS) prevails due to its feasibility in engineering practice. Unsteady flow simulations under normal operating conditions were developed first [33–35], and the results showed that for the monitoring point closest to the runner, maximum pressure amplitude was observed for the pressure fluctuation component of BPF. This phenomenon indicated the strong influence of the potential effect in the interactions between the guide vanes and the rotating runner blades. This component decreases rapidly backward to the stay vanes.

Studies on the off-design operating conditions to investigate different feature of pressure fluctuations have also been carried out. Yan et al. [36] simulated the hydrodynamics in a pump-turbine under off-design operating conditions in turbine mode, and found that the low frequency components appeared under runaway and low discharge conditions. Widmer et al. [37] performed similar simulations and found that during unsteady vortex formation, the vortices in the runner and the vaneless space

fluctuated simultaneously, and induced in-phase pressure fluctuations in the vaneless space. Yin et al. [38] predicted the pressure fluctuations under low partial flow rate conditions of pump mode, and suggested that the low frequency pressure fluctuations are problematic.

In engineering practice, one of the primary methods used for improving the stability of the start-up process caused by S-shaped characteristics involves using *Misaligned Guide Vanes* (MGV). In this method a few (usually one or two) pairs of guide vanes are pre-opened to a relatively large opening. Fig. 10 illustrates the comparison of guide vane opening distribution between using MGV and synchronous guide vanes. Xiao et al. [39] simulated the unsteady flow within the entire flow passage of a pump-turbine with MGV. Three arrangements of MGV of different opening angles were chosen to analyze the influence of MGV on the pressure pulsations. It was found that by using the MGV, the amplitudes of the pressure fluctuations close to the synchronous opening guide vanes were reduced significantly. Liu et al. [40] and Liu et al. [41] found that the pressure fluctuations close to the misaligned opening guide vanes have been greatly increased by adopting the MGV schemes. The use of MGV destroyed the more uniform flow distributions in the runner and guide vanes, which causes disturbances in certain flow passages.

Other attempts have been made to optimize the unit with respect to the pressure fluctuations. Casartelli et al. [42] proposed a simplified CFD model to study pressure fluctuations. The advantage of this approach is its cheaper computational cost, while on the other hand

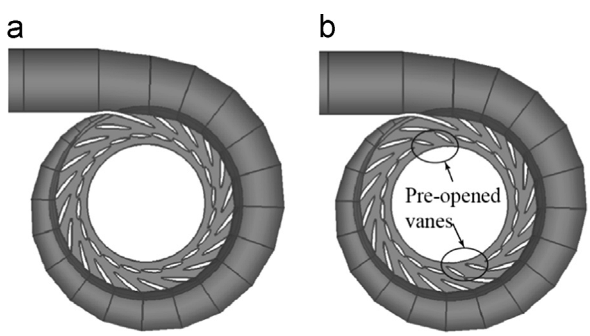

Fig. 10. Openings of guide vanes. (a) Synchronous guide vanes abs (b) misaligned guide vanes (MGV).

a much stiffer system behavior of pressure fluctuations was obtained. Ran et al. [43] performed the hydraulic runner optimization using the CFD method by changing the blade parameters at the exit to reduce the intensive rotor–stator interaction. The flow pattern and pressure fluctuation were both improved. It was expected that the flow instability of the pump-turbine could be mitigated.

Some numerical studies have attempted to consider the effects of compressibility and the existence of cavitation [44–46]. However, further investigation is needed for conclusive statements.

Sun et al. [47] carried out a study on the distribution of pressure fluctuations (BPF and its harmonics) of a prototype pump-turbine, both in circumferential direction, and along the flow in pump mode. It was determined that the two most important frequencies (BPF and 3BPF, since the numbers of the guide vanes and runner blades are 20 and 7, respectively) caused by RSI in the studied prototype pump-turbine showed totally different distributions along the circumferential direction and the flow path direction, as shown in Fig. 11. This indicated that different components of RSI (different interaction modes) might performed differently in the pump-turbine unit, which should draw more attention in future investigations. Guo et al. [48] carried out a similar investigation on a model pump-turbine at pump mode under its low head conditions. Different distributions of the pressure fluctuation components were also reported.

4. Relationship between vibrations and pressure fluctuations

The relationship between the vibrations and pressure fluctuations has been extensively discussed. Tanaka [20] has shown that the excitation force of vibration is developed by pressure fluctuations in vaneless space, which is induced by the interference of blade cascades. In Yixing power station, strong vibrations of guide vanes were experienced during the pump mode start-up and the turbine mode over-speed and double-unit trip tests, because of the large pressure fluctuations in the vaneless space [49]. It was concluded that the pressure fluctuations in vaneless space could cause stress and vibration of the pump-turbines.

On the other hand, the vibration of the guide vanes and the runner can also influence the pressure fluctuations. Roth et al. [50] from *École Polytechnique Fédérale de Lausanne* (EPFL) investigated the fluid-structure coupling in the guide vanes cascade of a pump-turbine scale model. Roth's work focused on the advanced instrumentation used to acquire reliable and complete fluid-structure

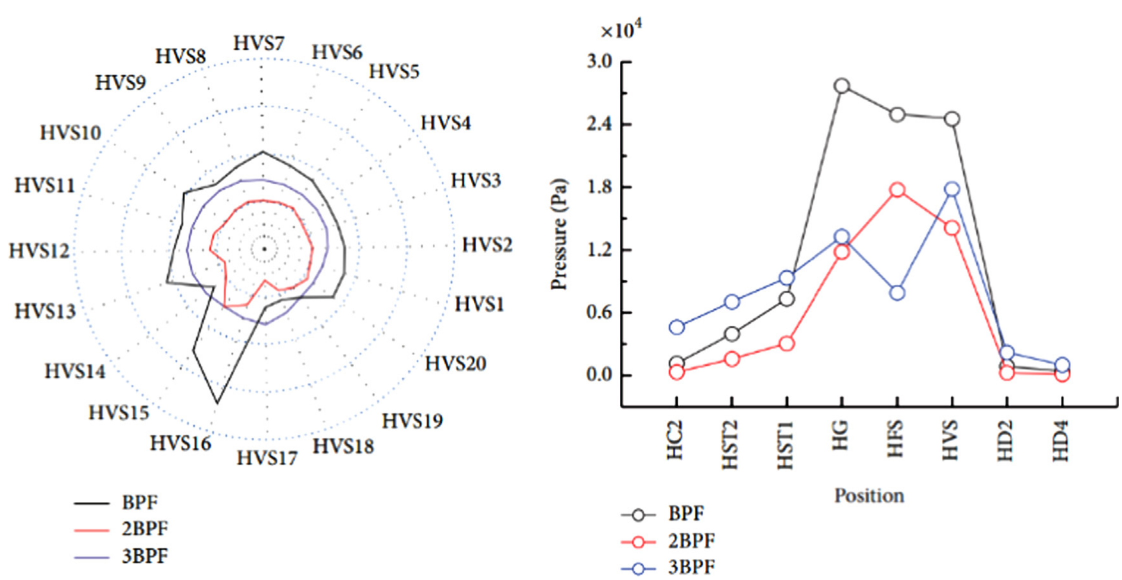

Fig. 11. Distribution of pressure fluctuations components of BPF, 2BPF, and 3BPF at different positions in circumferential position (left) and along a flow path (right) [47].

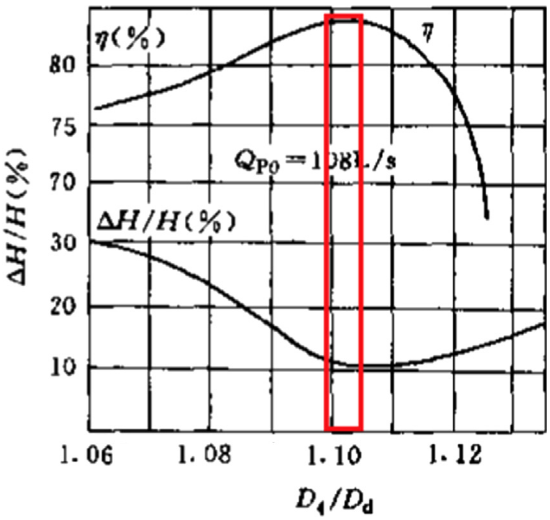

Fig. 12. Pressure fluctuations with different D_0/D_d [27].

coupling results, e.g., semi-conductor strain gauges, laser vibrometers, Piezo-resistive pressure sensors, and a non-intrusive impulse excitation system. The results showed that guide vane vibrations locally influenced the fluid pressure fluctuation in vaneless gap. The fundamental RSI frequency predominated in pressure signals in the proximity of vibrating structures with low amplitude, while the 1st harmonic of the vibration frequency was dominant compared to the fundamental frequency when close to a higher amplitude vibration structure.

In another article by Roth et al. [51], the relationship between the amplitude of the pressure fluctuations and vibrations was investigated. The amplitude of vibrating guide vanes was shown to have a direct impact on the amplitude of the pressure fluctuations in the upstream and downstream regions. The pressure fluctuations closer to the vibrating structures had much higher amplitude than those far away from the vibrating structure.

Roth et al. [52] also investigated the detailed influence of the guide vane vibrations on the pressure fluctuations in the vaneless space. The amplitude of the pressure fluctuations in the vaneless space between the rotor and stator has generally been shown to increase linearly with the operating specific energy, subjecting the pump-turbine components to high stresses at high head operations. The study showed that the guide vanes' vibratory properties might have a strong influence on the amplitude of the pressure fluctuations, due to the RSI. Thus, their amplitude might locally drop by 50% closer to the guide vanes reaching resonance. The authors of this paper have concluded that the resulting phase shift between their vibrating motions might be the reason of the pressure fluctuation amplitude attenuation.

5. Parameters influencing pressure fluctuations

Some parameters of pump-turbines that influence pressure fluctuations, apart from operating conditions, can be summarized as follows.

5.1. The guide vanes opening and the distance of vaneless gap

Ref. [27] studied the influence of guide vane openings and the distance of the vaneless gap to pressure fluctuations in pump mode. Fig. 12 shows the results in which pressure fluctuations vary with different D_4/D_d, where D_4 denotes the inner diameter of guide vanes, and D_d denotes the diameter of the runner. The flow rate was kept as a constant in this research. The results of this

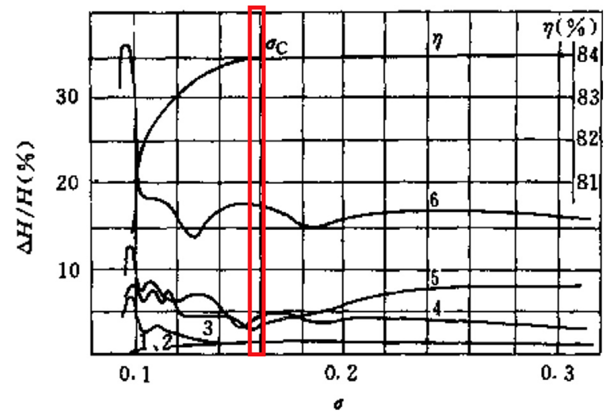

Fig. 13. Pressure fluctuations in vaneless space with different cavitation conditions [27].

research showed that when D_4/D_d was 1.1, the pressure fluctuations in vaneless space were lowest and the efficiency was highest at a certain flow rate operating point.

Another study on the influence of D_0/D_d has also been carried out, in which D_0 was the pitch diameter of guide vanes opening [53]. It was shown that when the flow rate was set as constant, smaller D_0/D_d caused higher pressure fluctuations in the vaneless space in pump mode, and lower pressure fluctuations in turbine mode.

5.2. Cavitation condition

Ref. [27] also studied the influence of cavitation conditions on pressure fluctuations in pump mode. As shown in Fig. 13, pressure fluctuations' amplitudes increased when the cavitation coefficient decreased. When the cavitation coefficient decreased to the critical cavitation coefficient number, the pressure fluctuations were 30–40% larger than at normal operating points.

5.3. The height of guide vanes

The height of guide vanes also influences pressure fluctuations in turbine mode [28]. The researchers performed experiments in three different pump-turbine models. The three models were different in terms of the height and the thickness of the guide vanes. The results showed that the thickness of guide vanes had little influence on pressure fluctuations. However, after increasing the guide vanes' height by 40%, the amplitude of pressure fluctuations in vaneless space was reduced by 20–30%. The pressure fluctuations in the draft tube remained constant when the height and thickness of guide vanes changed.

5.4. Application of MGV

The purpose of using MGV in PSPS is to improve the start-up instability in "S" characteristics of pump-turbines. However, it could also influence the pressure fluctuations in vaneless space. Xiao et al. [39] found that the pressure fluctuations in vaneless space close to synchronous opening guide vanes reduced significantly, while researchers in [39,40] found that the pressure fluctuations close to the misaligned opening guide vanes increased.

5.5. Parameters of runner blades

Ref. [43] studied the ways in which the exit angle and wrap angle of runner blades influence pressure fluctuations in vaneless space. Two runners with different blades exit angle and wrap

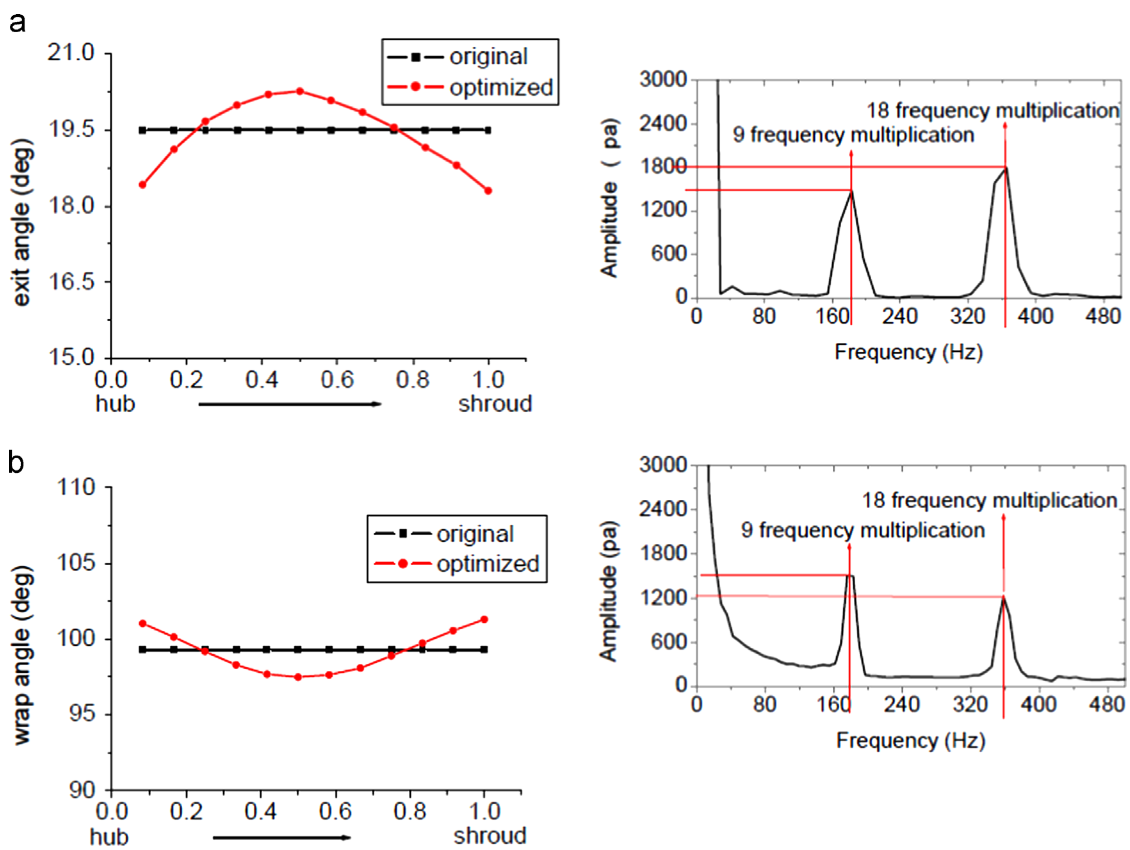

Fig. 14. Pressure fluctuations with different parameters of runner blades [43].

angle were compared in this research. As shown in Fig. 14, the results showed that using of twisted blades helped to reduce the pressure fluctuations.

5.6. Other parameters

It was argued that the using of splitter guide vanes [54] and splitter runner blades [55] helped reducing the pressure fluctuations. There are also other parameters that influenced the pressure fluctuations, such as the elongated body in the draft tube. However, the mechanism and principle of these parameters have not been verified.

6. Precautions and countermeasures

As previously discussed, different combinations of Z_g and Z_r causes different mode shapes of vibrations. Resonance between the pressure fluctuations and vibration modes of the runner must be avoided, especially with the low order of the runner harmonic k. Hence it is essential to select the combination of Z_g and Z_r carefully at the design stage [56]. Combinations of $Z_g=20$ and $Z_r=7$, and $Z_g=20$ and $Z_r=9$ have been frequently used in recent Chinese PSPS installations. Furthermore, discussions in Section 5 suggest the necessity of considerations regarding the influence of distance of vaneless gap, the height of the guide vanes, and contour of the runner blades (e.g., skewed blade tips [56]).

Since pressure fluctuations are usually more intense in the transient processes than under normal operating conditions, precautions need to be taken regarding the control of these processes. An experimental study of the start-up process in pump mode suggests that by opening the guide vanes before the complete evacuation of compressed air, large pressure fluctuations can be avoided [57]. Experimental tests indicated that the guide vanes opening procedure during start-up process in turbine mode influenced the transient hydraulic instability [58]. Cepa [59] reported that by utilizing an upstream biplane butterfly valve (BV) as an added artificial head, the turbine start-up of the unit was effectively improved. Although with the BV opening setup in his study no definite improvements were shown with regard to pressure fluctuations, further investigation of this procedure are encouraged.

7. Conclusions

An extensive literature review on the research of pressure fluctuations in the vaneless space of high-head pump-turbines has been carried out and the following conclusions have been drawn:

a. It has been observed in engineering practice that pressure fluctuations in vaneless space are the most detrimental hydraulic instabilities in high-head pump-turbines, caused primarily by rotor–stator interaction (RSI).
b. Through hydroacoustic and CFD analyses, the frequencies of main components of pressure fluctuations in vaneless space could be predicted, i.e., the blade passing frequency (BPF) and its harmonics if observed in stationery coordinates. The corresponding vibration/pressure fluctuation modes with k diametrical nodes can be specified through an analytical model of potential interaction.
c. There is a strong correlation between vibrations and pressure fluctuations in vaneless space, which deserves more investigation with regard to flow–structural interaction.

d. Geometric and operating parameters of the unit can strongly influence the pressure fluctuations in vaneless space.
e. Precautions must be taken in the design stage, as well as during transient process control, to reduce the damage of pressure fluctuations in vaneless space.

Extensive study is still required to develop a full understanding of the characteristics and the influence factors of the pressure fluctuations in vaneless space, giving particular consideration to the viscous interactions, the influence of the liquid compressibility and cavitation.

Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this article.

Acknowledgement

The authors would like to thank National Natural Science Foundation of China (no. 51076077) and National Science and Technology Ministry of China (ID: 2011BAF03B01) for their financial supports.

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